



Modeled and Measured Dynamics of a Composite Beam with Periodically Varying Foam Core

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Abstract

The dynamics of a sandwich beam with carbon fiber composite facesheets and foam core with periodic variations in material properties are studied. The purpose of the study is to compare finite element predictions with experimental measurements on fabricated beam specimens. For the study, three beams were fabricated: one with a compliant foam core, a second with a stiffer core, and a third with the two cores alternating along the length of the beam to create a periodic variation in properties. This periodic variation produces a bandgap in the frequency domain where vibrational energy does not readily propagate along the length of the beam. Mode shapes and natural frequencies are compared, as well as frequency responses from point force input to velocity response at the opposite end of the beam.

1 Introduction

Periodic variations of the material or geometric properties of a structure can be used to tailor the dynamic response of the structure and create frequency regions where vibrational energy does not readily propagate. This well-known phenomenon is often called bandgap behavior and has been studied for many years [1–6]. A frequency bandgap can be beneficial for isolating localized disturbances in a structure by reducing the propagation of unwanted vibrations through the structure. This can be especially useful in helicopters, where mechanical excitation from the main rotor transmission couples into the airframe and then re-radiates into the passenger cabin as unwanted noise. A periodic structure with a bandgap tuned to the excitation frequencies of the main transmission could be used to reduce the vibrational energy that eventually radiates into the cabin. Examples of this approach applied to helicopters can be found in the literature [7–9].

Periodic property variations in two-dimensional structures such as plates offer additional design possibilities due to the directional dependence of wave propagation in such a structure [5]. For example, properties could be varied in such a way as to direct flexural energy away from or toward specific regions of the structure [10]. This could be useful for concentrating energy at a localized damper or isolating an attachment point from disturbances.

Various methods have been described for analyzing and designing periodic structures. Many methods rely on a unit cell description of a structure, where the unit cell is the basic repeating structural element that makes up the structure. A transfer matrix method is used to describe the dynamics of an assembly of unit cells, and wave propagation through the assembly is then expressed in terms of complex propagation constants [2, 11]. Waves can be characterized as either freely propagating or attenuated, based on the wave propagation constants. For relatively simple unit cells, analytical expressions can be derived for the dynamic stiffness, and an eigenvalue problem can then be solved to determine the propagation constants [6, 12]. This approach is sometimes referred to as a spectral element technique [13]. For unit cells with more complicated dynamics, a finite element model can be used to determine mass and stiffness matrices for the cell, from which wave propagation properties can be determined, in what is referred to as a wave/finite element method [14].

In the present work, a more computationally intensive design approach is used whereby

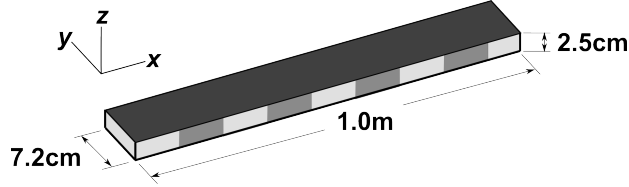


Figure 1. Nominal dimensions of beam.

the entire structure, in this case a beam, is meshed and analyzed using finite elements. The resulting model does not rely on division of the structure into unit cells, but requires that the structure be meshed using appropriate elements (e.g., 2D shell or 3D hexahedral elements). The disadvantage of this approach is that it results in a model that is much less compact and more computationally burdensome than the spectral and wave/finite element models. In addition, the relationship between unit cell dynamics and wave propagation behavior is more opaque than for the unit cell methods. The advantage of the approach, however, is a relaxation of the requirement that the structure be divided into unit cells. This allows for greater design freedom in satisfying an objective function such as isolation or concentration of energy in a 2D structure. An example of the complex property variations that result from relaxing the periodicity and unit cell requirement can be found in reference [10].

While the eventual goal of this work is the design of plates with complex arrangements of material properties, the present work concerns the design and testing of a periodic beam using readily available materials to create the property variations. Once the adequacy of the numerical model is established here by comparison with measured dynamics of fabricated beams, the approach can be applied with greater confidence to design plates. Natural frequencies, mode shapes, and frequency responses will be compared between the model and measured data.

Three beams were fabricated for this study. All three had carbon fiber facesheets and a core made from industrial grade structural foam (Rohacell®). This foam core material was chosen because it was readily available in a range of densities and stiffnesses. As a result, it was straightforward to create a sandwich beam with periodically alternating pieces of core material. Two of the beams were fabricated with uniform cores of either a stiff or compliant foam, while the third beam had a periodic combination of stiff and compliant foams.

The paper begins with a description of the beam geometry and material properties, followed by details of the finite element models. Fabrication of the beams is described next, followed by experimental measurements of the beam's dynamics. The modeled and measured dynamics are then compared, and adjustments made to the model to better match the measurements are quantified. The paper finishes with conclusions on the utility of the model.

2 Analysis

The nominal dimensions of the beams fabricated in this study are shown in figure 1. The beam consists of two composite laminate facesheets, each approximately 0.8 mm thick, that form a sandwich structure with a foam core that is approximately 2.5 cm thick. The facesheets were fabricated from and modeled as a laminate of Cytec Fiberite IM7/977-3

thickness	E_{11}	E_{22}	G_{12}	ν_{12}	ρ
0.127 mm	162.47 GPa	9.52 GPa	6.09 GPa	0.37	1585 kg/m ³

Table 1. Properties of IM7/977-3 cured prepreg ply. [15]

Foam	Elastic modulus, E (MPa)	Shear modulus, G (MPa)	Density, ρ (kg/m ³)	Poisson's ratio, ν (computed)
R31 ⁺	36	13	32	0.38
R110 ⁺	160	50	110	0.60
R110*	160	58	110	0.38

Table 2. Mechanical properties of industrial grade Rohacell[®] foam (⁺ - values from manufacturer; * - R110 with modified shear modulus)

carbon/epoxy, in a 6-layer layup with plies at angles of $60^\circ/0^\circ/-60^\circ/-60^\circ/0^\circ/60^\circ$, where 0° corresponds to the long axis of the beam. Published properties of a single cured facesheet ply are listed in table 1 [15].

The core material was Rohacell[®] industrial grade foam, which is a structural polymethacrylimide, closed-cell foam. This foam was available from the manufacturer in a range of densities and stiffnesses. A light, compliant foam, called R31, was selected as one core material, and a stiffer, denser foam, referred to as R110, was selected as the other core material. Mechanical properties of these two foams, provided by the manufacturer, are listed in the first two rows of table 2. The values in the table indicate the R110 foam is over 3 times denser and 4 times stiffer than the R31 foam. Assuming the foam is isotropic, Poisson's ratio is given by $\nu = E/2G - 1$, and the corresponding values are listed in the last column of the table. The computed value of 0.6 for the R110 foam is unrealistically high for an isotropic material, so the shear modulus used for that foam in the finite element models was increased by 16%, as shown in the last row of the table, to obtain a Poisson's ratio consistent with the R31 foam. In the future, material testing of the foam may be needed to determine a more accurate value of Poisson's ratio or more appropriate material model.

A finite element model was used to evaluate different core configurations and create a bandgap region. The model, implemented in MSC Nastran, consisted of plate elements (CQUAD4) for the facesheets and hex elements (CHEXA) for the core. An example of the finite element mesh of a beam with different core materials indicated by different colors is shown in figure 2. The nodes of the CQUAD4 facesheet elements and the outer CHEXA foam elements were coincident in the model, hence the facesheet elements are not apparent in the figure. The neutral axis of each CQUAD4 element was offset by half the laminate thickness in the model. The element sizes varied slightly depending on the beam configuration, but the hex elements were approximately 6 mm on a side along the length and through the thickness of the beam, and either 6 mm or 7.2 mm along the beam's width. This provided 4 elements through the thickness of the beam, and either 10 or 12 elements along the width.

Designing the bandgap consisted of adjusting the lengths of R31 and R110 foam that were repeated along the length of the beam. The goal was to create a gap in the natural frequencies of the beam, as computed from the finite element model, near 1 kHz. This gap in natural frequencies would then correspond to a bandgap in the frequency response. A

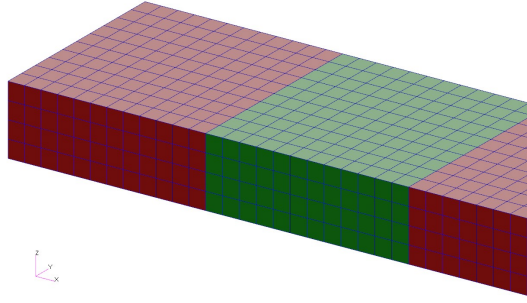


Figure 2. Typical mesh of solid elements used to model the sandwich beam.

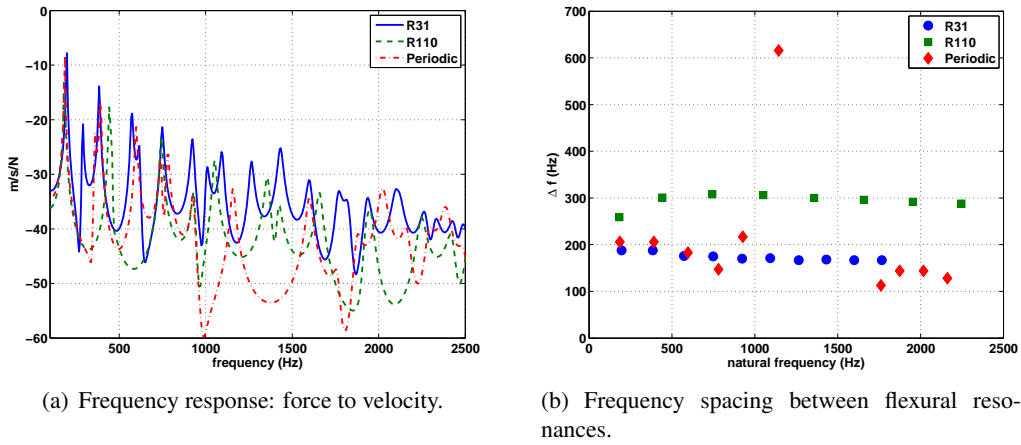


Figure 3. Finite element modeling results.

starting point for the design was to make the length of R31 close to the flexural wavelength at 1 kHz with the length of R110 some fraction of that wavelength. After numerous iterations, a final design was selected that consisted of equally sized pieces of R31 and R110, each 7.2 cm long, alternating along the length of the beam.

Computed frequency responses from a force excitation on one end of each beam to the out of plane (z -axis) velocity response at a point on the opposite end of the beam are shown in figure 3(a). The R31 and R110 beams correspond to beams with uniform core, while the periodic beam contains the alternating 7.2 cm pieces of R31 and R110 foam. The first resonance of all three beams was around 200 Hz, with subsequent resonances varying among the beams. The resonance peaks correspond to out of plane modes (flexural and torsional). A bandgap of ≈ 450 Hz is evident in the frequency response function (FRF) for the periodic beam between 1150 Hz and 1600 Hz.

The bandgap is even more pronounced if the spacing between natural frequencies of only the flexural modes is plotted for the three beams. Figure 3(b) shows flexural mode spacing for the three beams, where the marker at a given frequency denotes the frequency spacing, in Hertz, between the resonance at that frequency and the next highest flexural resonance. Resonances for the R110 beam are spaced approximately 300 Hz apart, while those for the R31 beam are spaced approximately 180 Hz apart. The bandgap in the periodic

beam name	total mass as fabricated	adhesive mass	initial FE model mass	updated FE model mass
R31	261	24	231	264
R110	428	24	369	423
periodic	345	28	301	345

Table 3. Mass (g) of fabricated and modeled beams.

beam is clearly denoted by the marker at 1150 Hz, which indicates that the next flexural resonance does not occur until 1760 Hz.

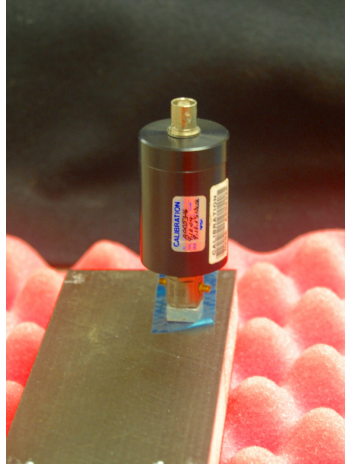
3 Fabrication

Three sandwich beams were fabricated to match the beams analyzed using finite elements. A 107 cm by 81 cm panel was fabricated from Cytec Fiberite IM7/977-3 carbon/epoxy unidirectional prepreg using the manufacturer’s recommended cure cycle. The laminate consisted of a 6-layer layup with plies at angles of $60^\circ/0^\circ/-60^\circ/-60^\circ/0^\circ/60^\circ$, where 0° corresponds to the long axis of the beam. The cured panel was then machined into 9.5 cm by 102 cm panels. These smaller panels were then secondarily bonded to either the R31, R110, or alternating sections of the two foams using FM-330-2 adhesive film. This film was also used to bond the alternating foam core pieces together. The facesheets were lightly sanded and solvent wiped prior to bonding. The sandwich panels were then bonded at 120°C for one hour with vacuum bag pressure only. Fully cured sandwich panels were then machined to the final size of 1 m by 7.2 cm for testing.

The beams were weighed after fabrication for comparison with weights computed from the finite element models. The fabricated and modeled masses are listed in table 3. The initial FE masses were computed using the facesheet and core densities listed in tables 1 and 2. The FE model neglected the mass of adhesive used to bond the facesheets to the core and bond the periodic cores together; the estimated mass of this adhesive is listed in table 3. To match the masses of the fabricated beams and account for the mass of the adhesive, the densities of the core and facesheet were increased by 14% in the FE models. The resulting updated model masses are listed in the last column of table 3.

4 Experimental Setup

Each beam was excited by a shaker and corresponding acceleration responses were measured in order to compare the responses with finite element predictions. The beams were placed on polyurethane foam to approximate free boundary conditions (i.e., no constraints on beam motion due to the supports). A Wilcoxon F5B shaker and associated Wilcoxon Z11 impedance head were attached vertically near one corner of the beam’s facesheet, as shown in figure 4(a). The shaker location was approximately 1.9 cm from the long edge and 4.4 cm in from the short edge of each beam. The shaker was attached by bonding a small aluminum nut, threaded onto the Z11 impedance head, to a piece of flashbreaker tape on the beam using a cyanoacrylate glue. Accelerations were measured at 20 locations along each beam using Endevco[®] 2250A accelerometers (0.4 g each), indicated by the white dots in



(a) Shaker in vertical orientation attached to facesheet of beam.



(b) Accelerometer locations (white dots) on facesheet excited by shaker.

Figure 4. Shaker and accelerometer configuration for testing.

figure 4(b). Accelerometers were attached to the beam using accelerometer wax. To minimize mass loading of each beam by the sensors, the response data were collected using only four accelerometers at a time, requiring five consecutive runs for the 20 measurement locations.

It should be noted that the accelerometers and impedance head used to make the measurements were not calibrated using NIST-traceable standards. Instead, scale factors were computed for the accelerometers and impedance head using an accelerometer calibrator (PCB[®] model 394C06) that produced nominally 1 g of acceleration at 159.2 Hz. The scale factors ensured the sensors responded with uniform magnitude to the same excitation, but the absolute level of that response cannot be traced to a calibrated standard. For this reason, the results here are discussed in terms of relative levels and computed quantities that are not dependent on absolute levels.

The coherence between the shaker force and the acceleration measured at the opposite end of the beam is shown in figure 5(a) for the three beams. The coherence was generally excellent away from resonances and above 100 Hz. Coherence values at the other accelerometer locations closely resembled the coherence shown in figure 5(a).

The uncalibrated power input to the beam by the shaker is shown in figure 5(b), in dB relative to the maximum power input. The curves indicate that the power input at 2000 Hz was nearly 40 dB below the maximum power input near 150 Hz for all three beams. Although the coherence from shaker to sensors remained high above 1500 Hz, the combination of reduced power input and shorter structural wavelength made it increasingly difficult to interpret the beam's response in terms of mode shapes above about 1500 Hz. In some

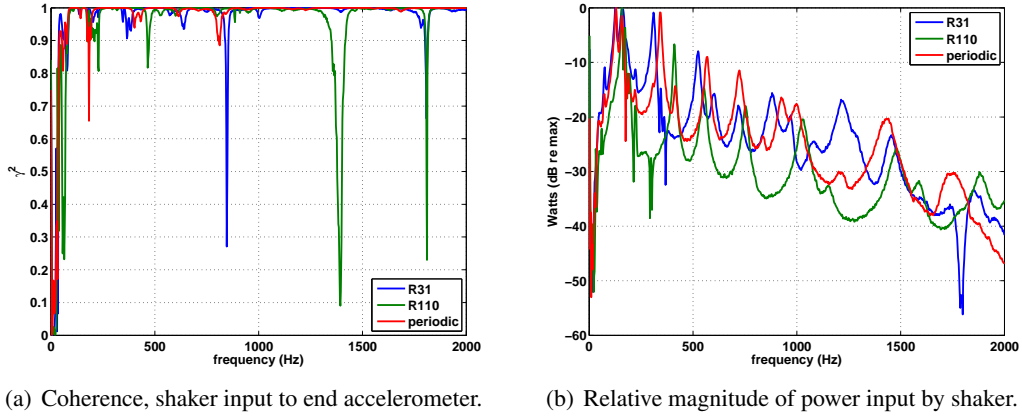


Figure 5. Measured coherence and input power.

cases, the response of all three beams above 1500 Hz was localized near the shaker attachment point. This made it difficult to interpret the beam's response in terms of modes, and difficult to judge the upper frequency end of the bandgap in the periodic core beam.

The accelerometer responses were post-processed to determine natural frequencies and associated mode shapes of the beams. These measured frequencies and mode shapes were compared with the finite element predictions in order to guide changes to material properties in the finite element models. The baseline material properties used in the model are listed in tables 1 and 2, although as discussed earlier, the densities of the foam and facesheet were increased by 14% so the modeled masses matched the fabricated masses.

Modeled and measured natural frequencies are compared in figure 6 for the first five flexural modes and first two torsional modes. The solid curves with circles correspond to flexural modes and the dashed curves with x's indicate torsional modes. Perfect agreement occurs when the curves coincide with the dashed line at 45° . In figure 6(a), the modeled natural frequencies were computed using the baseline material properties for the facesheet and core, including the 14% density increase. In figure 6(b), the material properties used in the model were adjusted to obtain better agreement with the measured frequencies. Specifically, the moduli of the facesheets (E_{11} , E_{22} and G_{12}) and the elastic modulus of the R110 core were increased 40% over their baseline values, and the elastic modulus of the R31 core was increased by 15%. The result is better agreement between the modeled and measured flexural modes, although the torsional modes are still underpredicted by the models.

There is no obvious physical justification for increasing the moduli, other than to obtain better agreement with the measured natural frequencies. For future modeling efforts, it would be worthwhile to measure the mechanical properties of the structural components, such as the core, facesheets, and adhesive, instead of using published property values.

Figures 7 and 8 show good agreement between predicted mode shapes and measured operating deflection shapes. Although the measurements are coarsely spaced relative to the fine finite element grid, the response data for these two modes agree well with the modeled mode shapes. In particular, the operating deflection shape in figure 7(b) displays the same asymmetry seen in the mode shape in figure 7(a). Similarly, the deflection shape at 1152 Hz shown in figure 8(b) shows a decaying behavior seen in the mode shape prediction

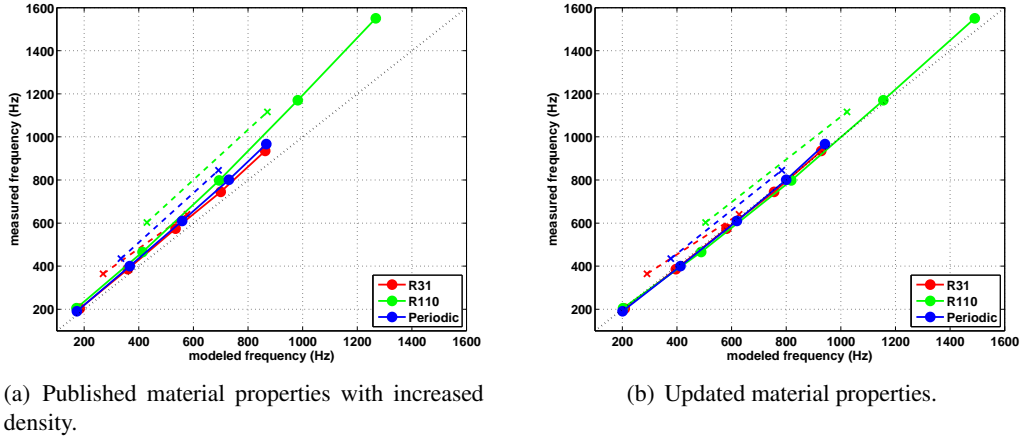


Figure 6. Comparison between modeled and measured natural frequencies.

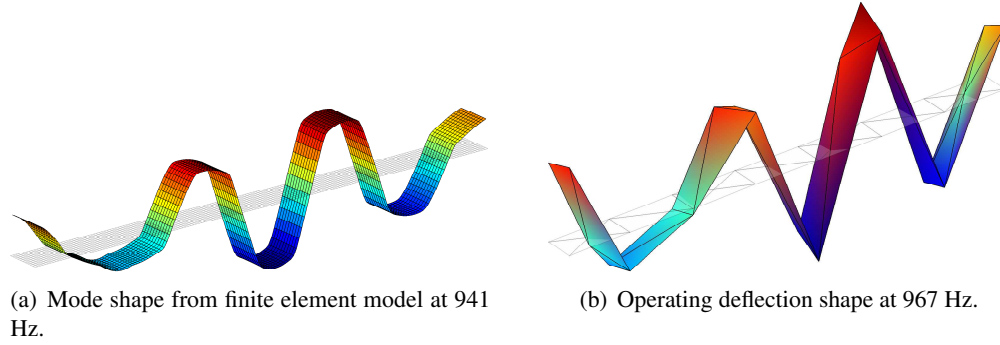


Figure 7. Comparison of beam shapes near 950 Hz.

in figure 8(a).

Measured frequency responses were averaged in order to highlight the presence of a frequency bandgap in the flexural response of the beam. The complex frequency responses for the two accelerometers mounted on the far end of the beam, away from the shaker attachment, were averaged together. The averaging should emphasize in-phase motions of the two accelerometers due to a flexural response, and minimize out-of-phase motion due to a torsional response. Similar averaging was applied to the modeled frequency responses of the 10 or 12 nodes (depending on the particular beam mesh) on the far end of the beam.

The resulting averaged frequency responses for the three beams, normalized relative to their maximum values, are shown in figure 9. In general, the measurements and predictions agree well in terms of peak location and decrement of peak amplitude with frequency, up to about 1500 Hz. Above that frequency, the agreement is not as good, and may be due to the apparent localized excitation of the beam by the shaker that was noted earlier.

Although the agreement is generally good below 1500 Hz, some discrepancies are

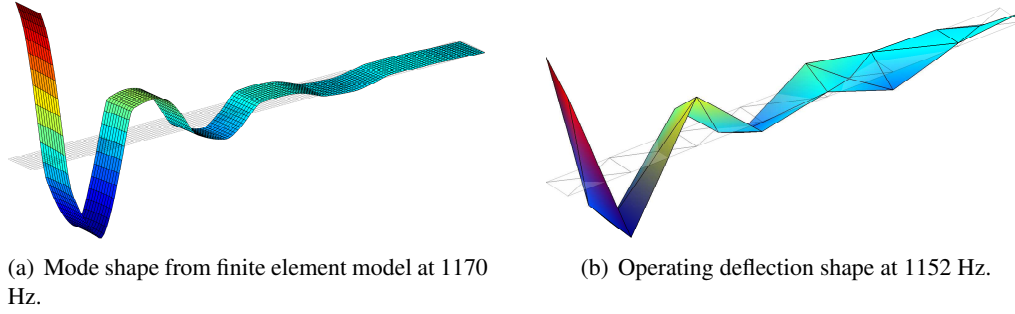


Figure 8. Comparison of beam shapes near 1150 Hz.

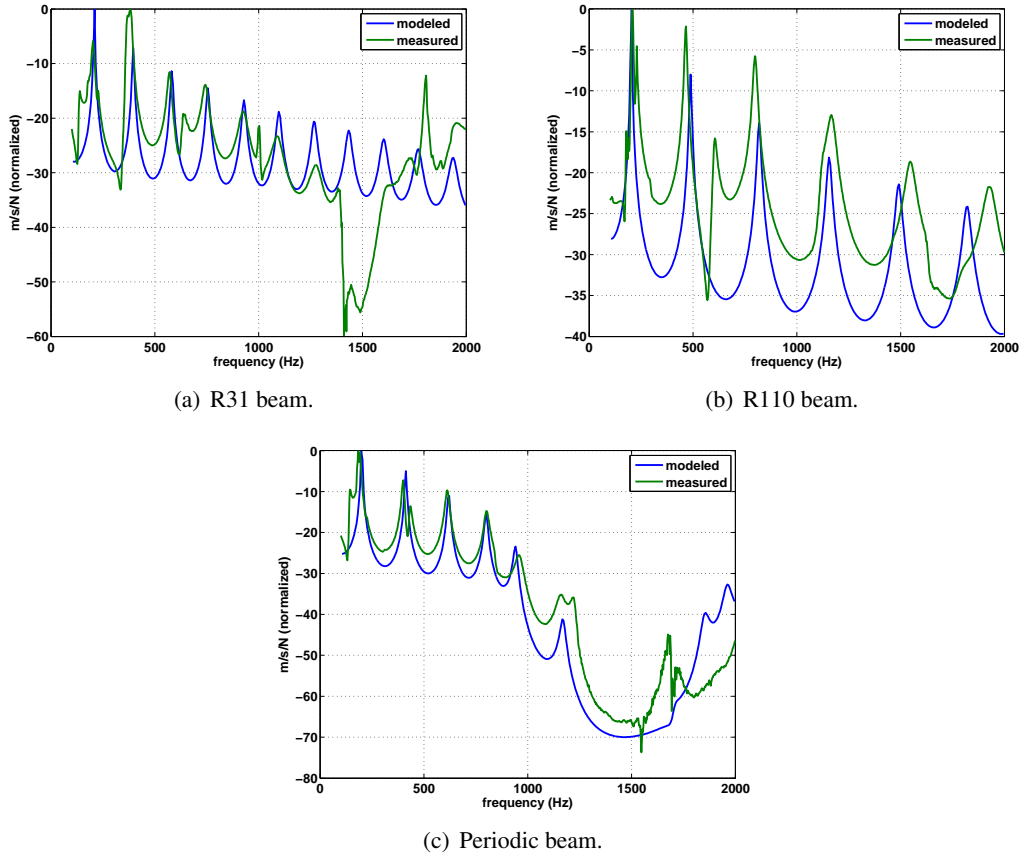


Figure 9. Averaged frequency responses between an input force and the response of the opposite end of the beam.

prominent. In figure 9(a), the measured frequency response for the R31 beam appears to show a bandgap effect from just above 1400 Hz to 1600 Hz. However, this dip can be traced to two effects: the first is a torsional mode at 1410 Hz, which creates nearly equal and oppositely-phased responses at the two accelerometers. This symmetric response was produced even though the exciting force was offset from the beam's centerline. The average of the accelerometers' responses to this symmetric response is numerically very small. In addition, the beam's response is very low from 1450 to 1600 Hz, presumably due to poor excitation from the shaker. In the R110 data, the peak near 600 Hz in the measured but not the modeled data is due to a torsional mode at 610 Hz with a strongly asymmetric response at the two accelerometers, thereby creating a peak in the averaged frequency response.

The data corresponding to the periodic beam, in figure 9(c), contains a significant bandgap of reduced response between ≈ 1000 and 1700 Hz. Some of this response reduction, particularly above 1300 Hz, can be attributed to poor excitation of the beam by the shaker. However, the balance of the response reduction, from 1000 to 1300 Hz, occurs because the periodic core both creates a modal response, shown in figures 8(a) and 8(b), that allows high displacements at the driven end and minimal displacements at the opposite end, and creates a gap in natural frequencies above 1000 Hz. The result is a reduction in the magnitude of the frequency response from shaker input to velocity response at the opposite end of the beam.

5 Conclusions

Sandwich beams with composite facesheets and foam cores were fabricated and dynamically tested in order to evaluate the accuracy of finite element models of the beams. Three beams were fabricated: one with a light, compliant foam core, a second with stiffer, denser foam core, and a third with alternating pieces of the two foam cores. The alternating, or periodic, core was designed to create a frequency bandgap where flexural waves did not propagate along the beam. The finite element models used shell elements to model the facesheets and hexahedral elements to model the foam core.

Comparisons between predicted and measured natural frequencies for the three beams were generally good after material properties used in the models (densities and stiffnesses) were updated. Specifically, densities for the facesheet and foam cores were increased 14% to match the fabricated beam masses. In addition, the moduli of the facesheets and the heavier core were increased by 40% over published values, and the modulus of the lighter core was increased by 15% over its published value. With these new material properties, modeled and measured frequencies of the first five flexural modes of the beam agreed to within 6% of one another, while torsional modes agreed to within 20%. These frequency discrepancies, combined with the large increases made to the material moduli in the models, suggest that future models should rely on measured material properties of the beam components before they are bonded together. There should also be some effort to account for the mass of the adhesive used to bond the facesheets and periodic core pieces together. Modeled mode shapes and measured operating deflection shapes showed good agreement for the periodic beam, with both showing similar asymmetries due to the periodic core material.

The generally good agreement between measurement and predictions suggests that finite element models should be sufficiently accurate to further explore the approach of vary-

ing the core properties in order to tailor vibration response behavior of a beam or plate.

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